HIGH PERFORMANCE BEARING COMPARISON

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ABSTRACT

With the increasing demand for high performance gearboxes, larger, faster, more highly loaded bearings are requiring more oil and creating more heat than ever before. This means the lubrication systems must be larger to handle the increasing heat loads and oil demands. Offshore applications, in particular, are greatly affected due to space constraints and increased lubrication system size and cost. In an effort to reduce the oil flow and heat load requirements for the gearbox, experimental tests and field tests were performed with three different bearing designs. The designs were pressure dam, offset half, and tilting pad journal bearings. Data was acquired using a dedicated test rig that allows operation at and beyond design speeds and loads. Field test data was also collected from a full speed, full load string test of a turbo compressor drivetrain. This paper will present results of the experimental test data from these three bearings to assist in the selection of a design that will provide optimum performance for given operation conditions.

INTRODUCTION

The extraction, refining, and processing of oil and gas requires a variety of turbomachinery equipment, such as compressors, expanders, and generators. Much of this equipment relies on gearboxes to function at optimum speeds. Turbomachinery gearboxes are generally considered to be relatively efficient equipment; typically achieving measured efficiency values greater than 98 percent. Even at these efficiency levels, gearboxes in turbo-equipment strings, which commonly operate at or above 55,000 hp (40 MW), can have considerable losses. The losses on these units are comparable to the amount of power created by the engines of the world's fastest supercars.

The power consumed by the gearboxes is from mechanical shearing of oil and aerodynamic windage from gears, bearings and couplings. A large portion of the losses come from the bearings. Most of the lost power is converted into heat energy and dissipated with circulating oil. In order to absorb this heat, lubrication systems must be sized accordingly. Lubrication systems can often be an expensive and real estate intensive subsystem for turbomachines. Therefore low power losses and oil flows are very desirable traits for gearboxes. In an effort to make improvements in these areas, a variety of bearing designs needed to be tested for use in geared turbomachinery applications.

Significant work on research, testing, and analysis has been dedicated to studying rotordynamics and stability of journal bearing designs which will only be covered at a very high level in this paper. This paper will instead focus on a direct comparison of three competing bearing designs in the same gearbox application, with special interest given to oil consumption and power loss. The following is a brief overview and description of the bearing designs chosen for this test.

BEARING DESIGNS

Hydrodynamic journal bearings are currently the bearings of choice for most turbomachinery equipment including gearboxes. They possess several characteristics that are desirable such as high reliability and good rotordynamic damping characteristics. Several different designs of journal bearings are commonly utilized for gearboxes. The designs are all variations of a sliding bearing where a shaft journal slides on a thin film of oil. The design variations utilize different geometries and features in an effort to achieve rotordynamic stability and avoid sub-synchronous vibrations. The designs that were tested were pressure dam, offset halves, and tilting pad journal (TPJ) bearings.

Pressure Dam bearings are essentially a plain journal bearing with a pocket cut in one half that has an abrupt stop at some point in the shell, see Figure 1. Viscous and inertia effects results in a buildup of pressure at the dam, this localized high pressure area creates an artificial load on the shaft that helps stabilize the rotor, Wilcock and Booser (1957).

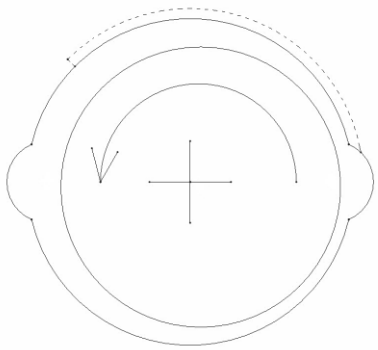

Figure 1 - Pressure Dam Journal Bearing

Offset half bearings utilize geometry where two bearing shell pads are not concentric to each other, see Figure 2. This creates a situation where oil is converged into a wedge helping stabilize the shaft in the bearing, Allaire and Flack (1981).

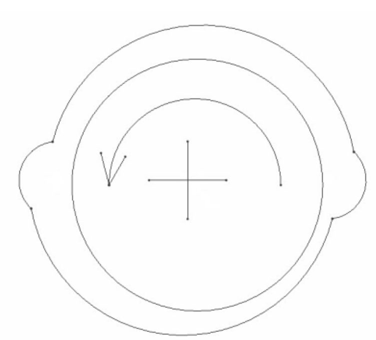

Figure 2 - Offset Halves Journal Bearing

Tilting Pad Journal Bearings are comprised of multiple pads that are supported by pivots, see Figure 3. The pads support the shaft and pivot independently from each other. The pads are typically machined at a larger diameter than they are assembled to create a converging oil film. The tilting pad design is inherently very stable.

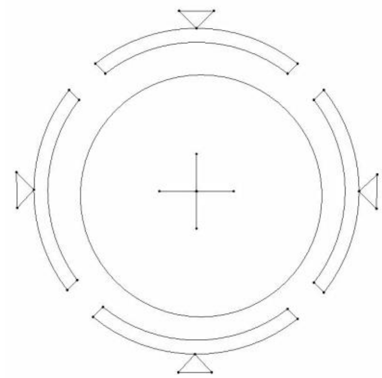

Figure 3 - Tilting Pad Journal Bearing

The three previously mentioned bearing designs are by no means an exhaustive list of the possible options, but are common in turbomachinery and tested for this application.

TESTING CONFIGURATION

The dedicated test rig consists of a shaft, a test bearing, support bearings, a load cell, a hydraulic cylinder, and a fabricated housing, see Figure 4 and Figure 5. The test bearing and support bearings have separate oil supplies and drains which allow for the oil flow to each bearing to be directly measured and the power loss to be calculated.

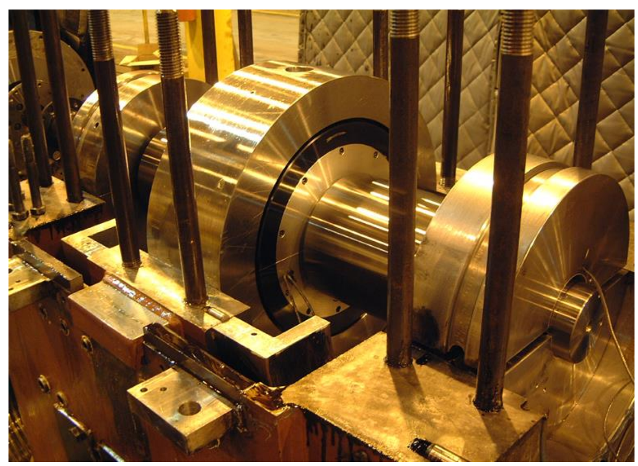

Figure 4 - Dedicated Test Rig with Cover Removed

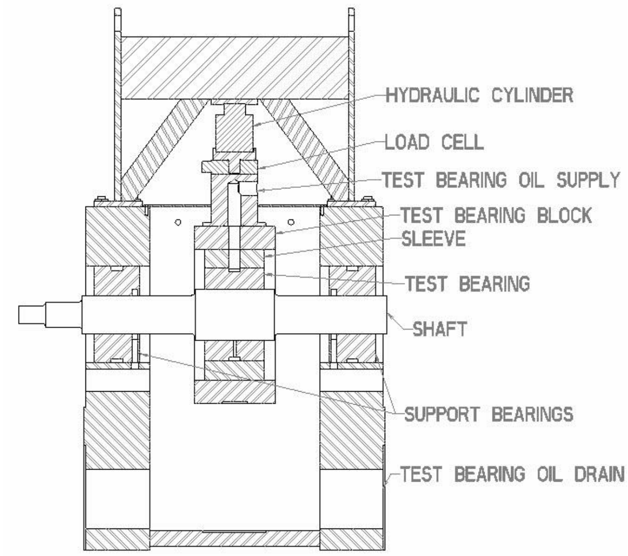

Figure 5 - Test Rig Diagram

The test bearing sits in the middle of the shaft and is straddled by the support bearings. The shaft is coupled to an electric motor and rotated to simulate gearbox operating conditions. The test rig allows a load to be applied to the floating bearing block using a hydraulic cylinder. The amount of load is monitored by using a load cell. The following items are continuously monitored and recorded periodically while testing:

- Shaft speed
- Hydraulic cylinder pressure
- Static load
- Test bearing metal temperatures
- Support bearing metal temperatures
- Oil inlet temperatures
- Oil outlet temperatures
- Oil pressure
- Ambient temperature

Dynamic bearing characteristics and shaft vibrations were not measured during testing.

TEST BEARINGS

Two different sizes of bearings were tested, 6 inch (152.4 mm) bore and 6.69 inch (170mm) bore. Pressure dam, offset half, and evacuated Tilting Pad Journal bearing designs with 6 inch (152.4 mm) bores were tested using the dedicated test rig. See Tables 1-3 for geometry details. The 6.69 inch size offset half and evacuated TPJ bearing designs were tested with the test rig and in a full load string test. Images of the tested bearings can be seen in Figures 6-8.

Table 1 - Pressure Dam Geometry

Pressure Dam		
Journal Diameter	(in)	6.000
	mm	152.40
Bearing Length	(in)	6.000
	mm	152.40
Radial Clearance	(in)	0.0105
	mm	0.2667
Pressure Dam Width	(in)	4
	mm	101.6
Pressure Dam Depth	(in)	0.012
	mm	0.3048
Pressure Dam Angle (Deg)		135

Table 2 - Offset Halves Geometry

Offset Halves		
Journal Diameter	(in)	6.000
	mm	152.40
Bearing Length	(in)	6.000
	mm	152.40
Vertical Bore	(in)	6.0165
	mm	152.82
Horizontal Bore	(in)	6.0105
	mm	152.67

Table 3 - TPJ Geometry

TPJ		
Number of Pads		4
Pad Offset		65%
Set Bore	(in)	6.0087
	mm	152.62
Pad Bore	(in)	6.012
	mm	152.70
Pre load	(in)	0.27
	mm	6.86
Bearing Clearance	(in)	0.009
	mm	0.229
Pad Clearance	(in)	0.0123
	mm	0.3124

All of these bearing designs incorporate highly thermal conductive copper chrome shell material and bypass cooling technology to allow for higher loading than comparable steel backed bearings, Nicholas (2003).

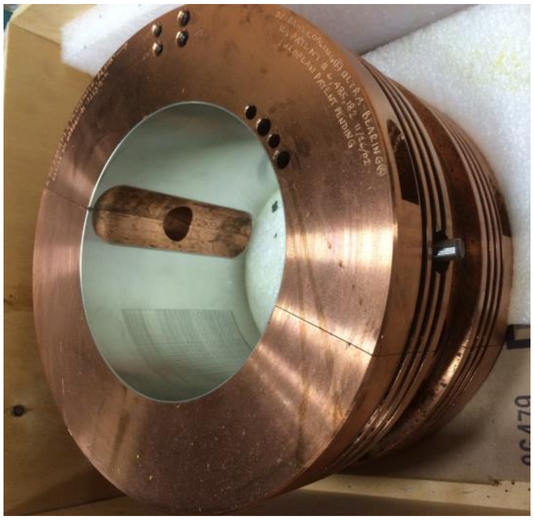

Figure 6 - Pressure Dam Bearing

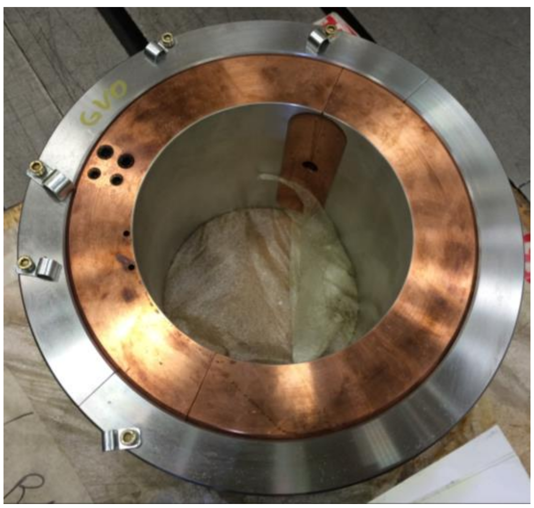

Figure 7 - Offset Halves Bearing

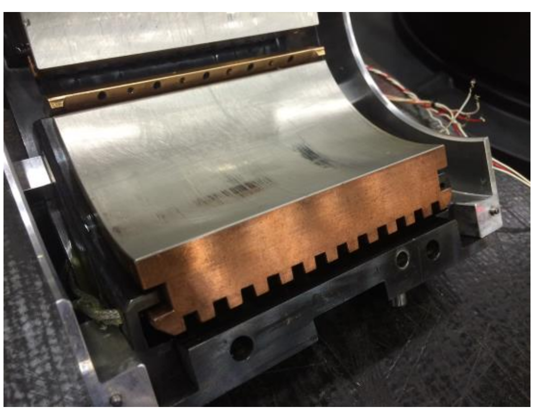

Figure 8 - Tilting Pad Journal Bearing

Bypass Cooling Flow Rates

Nicholas (2003) explains the bypass cooling flow applied behind chromium copper alloy tilting pads. This flow is about 10 percent of the flow used for lubrication. Figure 9 shows the equivalent bypass cooling arrangement for a fixed-geometry bearing. The flow assists convection over the surface area not contacting the bearing support. A bypass flow rate of 4 gpm (15.1 lpm) was chosen for all bearings tested in this application.

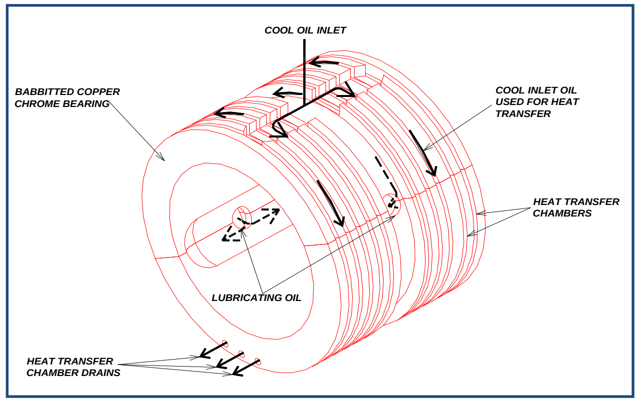

Figure 9 - Bypass Cooling for Fixed-Geometry Bearings

Bearing Lubrication Flow Rates

If a sufficient volumetric flow rate of lubricating oil is provided at the supply port, the fixed-geometry bearing takes the lubricant it needs based on the sliding speed, acting load, and the annular volume described by the bearing features and the shaft. The bearing assembly itself is the drain restriction. Analytical models suggest this flow is a relatively strong function of speed.

The tilting-pad bearing with an evacuated housing, as tested, has only the supply orifices for a flow restriction. The orifices are separated from and directed at the tilting pads. The bearing assembly cavity is at ambient pressure. The individual tilting pads naturally have a pressure profile when sliding, but they are at ambient pressure at their boundaries. Figure 10 shows part of a representative evacuated tilting-pad assembly. Lubricant flow is only a function of the supply pressure and orifice area in this case. This means that the orifices must be carefully designed for the most demanding operating condition. Nicholas, et al. (2008) discusses the steps used to determine the oil flow requirements. They are summarized with updates below:

1. Assume that the hot oil carryover from pad to pad equals the amount of oil that passes through the pad minimum film thickness.
2. Determine the minimum flow needed for a full film at the leading edge of the most loaded pad at full load and speed.

3. Determine the minimum flow needed for a full film at the leading edge of the pads near zero load at full speed.
4. Take the mean of steps 2 and 3 and multiply by the number of pads to get the minimum flow requirement.
5. Increase this flow as needed to satisfy metal temperature or drain temperature rise requirements.

This method results in partial starvation at the "unloaded" pads at all load conditions at full speed and partial starvation at all pads near zero load at full speed. This is a practical compromise that tests successfully for gearbox and geared system designs with variable loads. The conservative alternative of providing full flow at the leading pad edges near zero load and full speed is impractical in many cases. The tilting pad flow rate applied here could also be reduced further. The anticipated effect is an increase of the metal temperatures. The effect on power loss is uncertain, because the increased temperature rise is balanced by the reduced mass flow rate. Bearing stiffness and damping are expected to change as well, as the acting pad films are reduced. Such a study is of substantial interest on its own and is not pursued here.

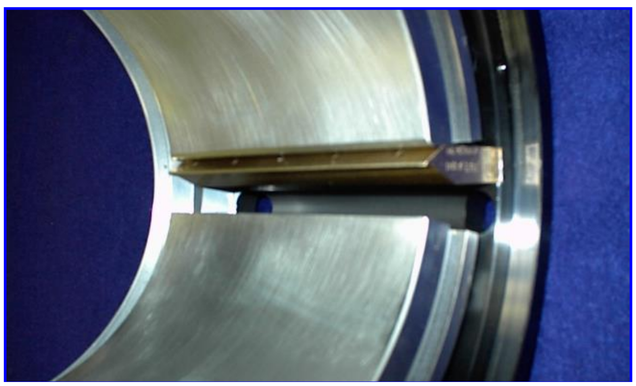

Figure 10 - Evacuated Tilting Pad Assembly

FORMULAS

To compare bearing results a few formulas are needed. Power loss of the bearings is calculated using the oil flow of the bearing and the temperature rise across the bearing. Equation (1) is a thermal power loss formula to calculate the losses in each bearing.

Equation 1 - Power Loss

$$Power\ Loss = \gamma * C_p * Q * (T_{drain} - T_{in})$$

γ - Weight per unit volume
C_p - Specific heat of oil
Q - Volumetric flow rate
T_{drain} - Drain temperature
T_{in} - Oil inlet temperature

In this test (γ*C_p) was approximated at 0.0817 hp/(gpm*°F).

Equation (2) was used to calculate the projected load of a bearing. Projected load is the force applied to the bearing divided by the projected cross-sectional area of the bearing.

Equation 2 - Projected Load

$$Lu = \frac{F}{D * W}$$

L_u - Projected load
F - Force applied to bearing
D - Journal diameter of bearing
W - Width of bearing

Shaft surface velocity can be calculated using Equation (3).

Equation 3 – Surface Velocity

$$Surface\ Velocity = \omega * \pi * D$$

ω - Shaft rotational speed (rpm)
D - Journal diameter

These are general forms of the equations that are suited for calculations in imperial or metric units. Common conversions maybe needed to match units used in the paper.

TEST PROCEDURE

Each bearing was tested at several speeds and loading conditions. Table 4 shows the speeds and loads for each bearing tested.

Table 4 - Bearing Testing Conditions

6 inch (152.4 mm) Bearings						
RPM	Surface Velocity		Projected Load		Hydraulic Ram Force	
	fps	m/s	PSI	MPa	lbf	N
8000	209	64	50	0.34	1800	8007
10000	262	80	200	1.38	7200	32027
12000	314	96	350	2.41	12600	56048
14000	367	112	500	3.45	18000	80068
16000	419	128	650	4.48	23400	104088
6.69 inch (170 mm) Bearings						
RPM	Surface Velocity		Projected Load		Hydraulic Ram Force	
	fps	m/s	PSI	MPa	lbf	N
10729	313	95	45	0.31	2000	8896
			248	1.71	11110	49420
			496	3.42	22220	98839

For each test the shaft was rotated at a nominal speed for two hours to allow the rig to be heat soaked prior to collection of any test data.

Initial test points for each bearing were gathered at the lowest speed and load conditions. Once the measured parameters stabilized, the load conditions were varied and data points were taken. Each loading condition was measured at the initial speed. The speed was then increased until all loading conditions were measured. This process continued until all data points were recorded or until the test bearing or support bearings reached 250°F (121°C). The shaft speed reached at least 14,000 rpm, which equates to a bearing surface velocity of 367 fps (111.9 m/s) and a load of 650 psi (4.48 MPa) for all test bearings. The maximum speeds and loads surpass typical operating conditions for hydrodynamic journal bearings in a gearbox. The 6.69 inch bearings were contract bearings and not dedicated test bearings and were only tested at 105 percent of designed operating speed. After all data points were recorded, a few points were rechecked to insure no significant hysteresis.

ANALYTICAL COMPARISONS FOR 6 INCH BEARINGS

A commercially available bearing analysis code was used for comparisons to tested flow, power loss, and maximum temperature. Multiple codes are not compared to the test data here. For background on bearing code variations, Kocur, et al. (2007) discuss the results from multiple codes attempting to model the same bearing design.

Pressure Dam Model

Figure 11 compares oil flow prediction to tests of the pressure dam bearing. The analytically predicted oil flow was very consistent with test data at low speeds, but the results started diverging at higher shaft speeds. The analytical model show a notable increase in flow as a function of shaft speed while the test data oil flows were nearly constant over the tested speed range. Power loss comparisons are shown in Figure 12 for 50 psi and 650 psi loading. Figure 13 plots temperatures vs. loading at 8,000 rpm, 12,000 rpm, and 14,000 rpm. Power loss predictions are slightly lower than tests, while temperature predictions are consistently higher than tests for speeds below 14,000 rpm. The 6" pressure dam bearing design is not recommended at 14,000 rpm (366 fps).

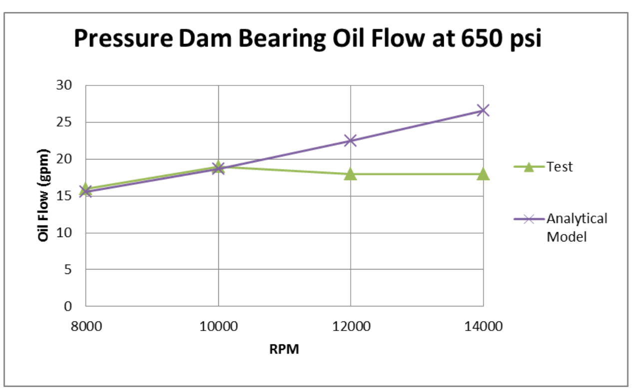

Figure 11 - Bearing Oil Flow vs Speed

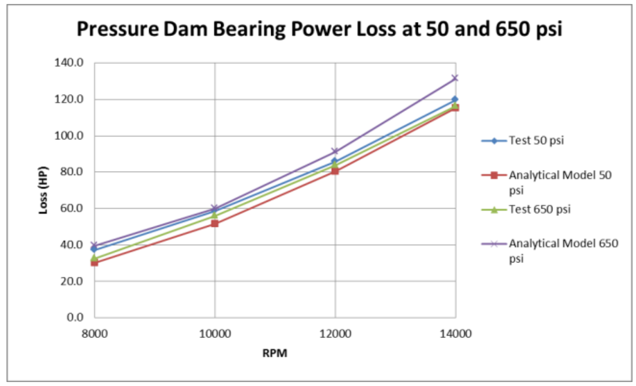

Figure 12 - Bearing Loss vs Speed 50 and 650 psi Loading

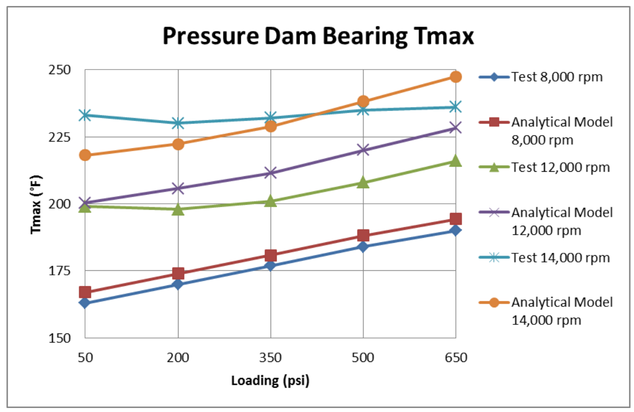

Figure 13 - Metal Temperature vs Loading

Offset Halves Model

Figure 14 compares oil flow prediction to tests of the offset halves bearing. The increase of flow with speed is again to be expected for the fixed geometry model. The test data on this bearing shows the flow has a lower correlation to speed than the analytical predictions. Power loss comparisons are shown in Figure 15 for 50 psi and 650 psi loading. Figure 16 plots temperatures vs. loading at 8,000 rpm, 12,000 rpm, and 16,000 rpm. The model fits the testing up to and including the limit of 16,000 rpm (419 fps).

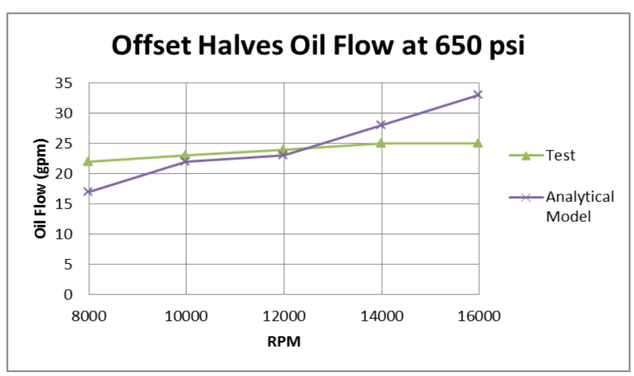

Figure 14 - Bearing Oil Flow vs Speed

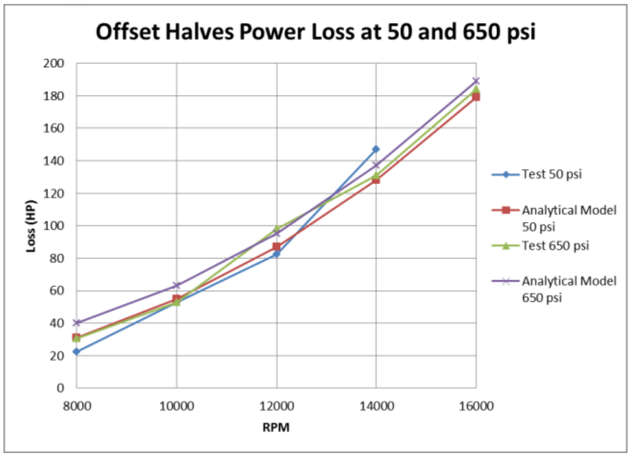

Figure 15 - Bearing Loss vs Speed50 and 650 psi Loading

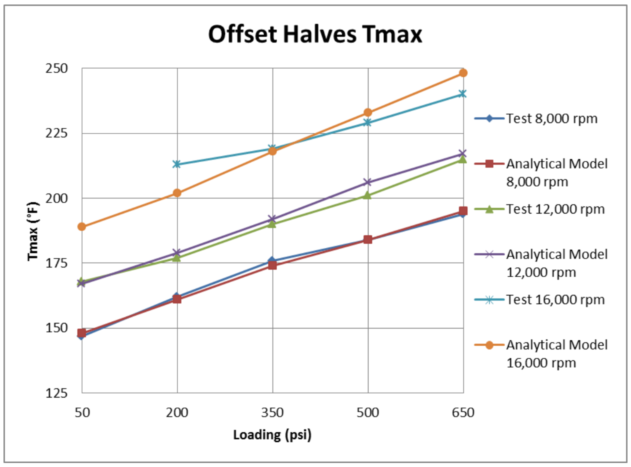

Figure 16 - Metal Temperature vs Loading

Tilting Pads Model

Flows for the tilting pads are only dependent on the restrictions of orifices in this design. Orifice calculations matched the tested flows here. Power loss comparisons are shown in Figure 17 for 50 psi and 650 psi loading. Figure 18 plots temperatures vs. loading at 8,000 rpm, 12,000 rpm, and 16,000 rpm. The model shows only partial agreement, as good temperature predictions throughout most of the test range are tempered by excessive power losses. Turbulence correction in the tilting pads model is one major influence on loss and temperature correlation. The critical Reynolds number is set to 950 in the analytical models. The bearings reach this value by 10,000 rpm for all load cases and by 8,000 rpm for the maximum loading.

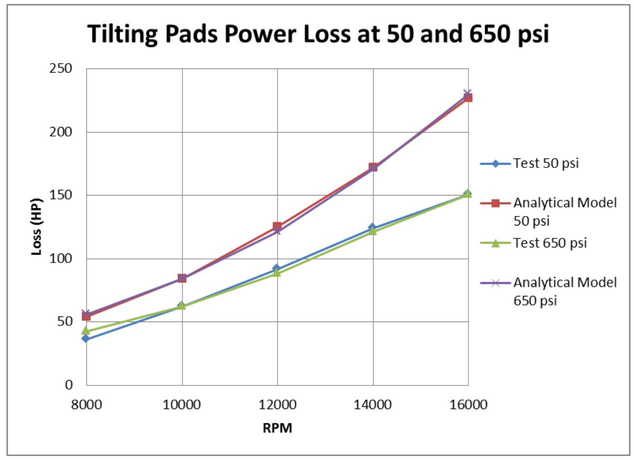

Figure 17 - Bearing Loss vs Speed 50 and 650 psi Loading

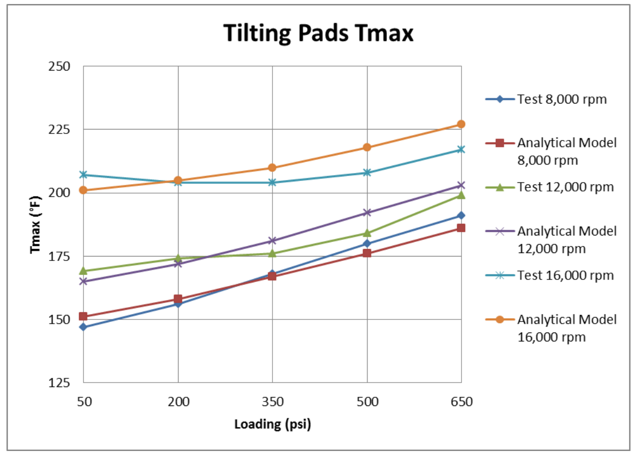

Figure 18 - Metal Temperature vs Loading

BEARING DESIGN COMPARISON

The previous results compared the analytical predictions to the test data for the various bearing designs. Direct comparisons of bearing test data for each bearing design were also reviewed.

RESULTS 6 (152.4 MM) INCH BEARINGS

Figure 19 shows the maximum oil flow observed for each speed of the bearing designs tested. The oil flow for the pressure dam and offset half bearings is considerably less than that of the TPJ.

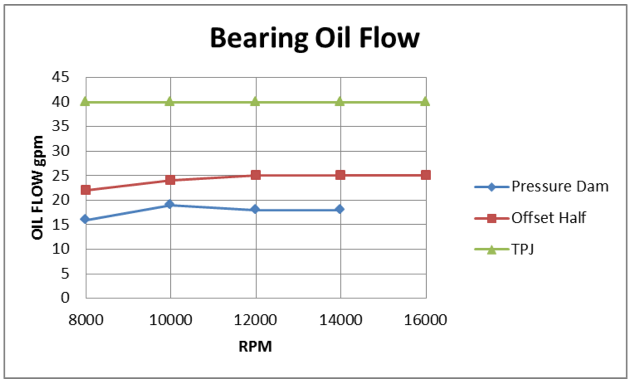

Figure 19 - Oil Flow of Test Bearings Over Tested Speed Range

Table 5 shows the power loss of the different bearing designs at the tested speed and load conditions.

Table 5 - Power Losses of Various Bearing Designs

RPM	Power Loss (hp)					
	Pressure Dam		Offset Halves		TPJ	
	Bearing Load (psi)		Bearing Load (psi)		Bearing Load (psi)	
	50	650	50	650	50	650
8000	29	33	22	31	36	42
10000	57	56	53	53	62	62
12000	84	84	82	98	92	88
14000	128	116	147	131	124	121
16000			***	184	150	150

*** - Data point was not obtained due to support bearing temperatures.

This table shows that load has a relatively small effect on the overall power loss of the bearings.

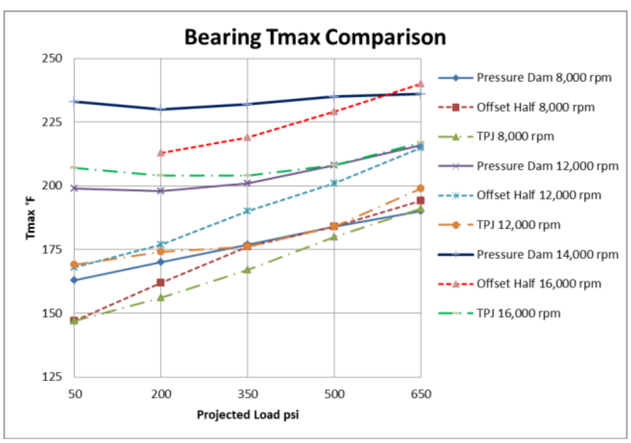

Figure 20 - Bearing Metal Temperature vs Projected Load

Figure 20 shows the bearing metal temperatures at the test conditions. The results show that as the surface velocity increases the bearing metal temperature also increases. The TPJ maintains the coolest bearing temperature of the three designs with the pressure dam measuring the highest temperatures.

Together, Figures 19 and 20 lead to the reasonable conclusion that the bearing with the highest lubrication flow rate has the lowest metal temperature. An interesting comparison was made by enforcing the offset halves bearing flow rate onto the tilting pad bearing analytical model. The result was roughly a 4% increase in metal temperature at 12,000 rpm and an 8% increase in temperature at 16,000 rpm. This indicates the tilting pad bearing though starved past 12,000 rpm may still have a temperature advantage over the offset halves bearing for the same flow rate. Such a prediction in turn needs to be verified by testing and may be of limited value when evacuated pad flow requirements are known. (API 613, 2003) standard allows a maximum projected load of 500 psi (3.45 MPa). If a reasonable temperature limit of 212 °F (100 °C) is used the pressure dam bearing is limited around 12,000 rpm (314 fps), the offset half 14,000 rpm (367 fps), and the TPJ 16,000 rpm (419 fps) based on this test data.

FULL LOAD STRING TEST DATA

The 6.69 inch (170mm) offset halves and TPJ bearings that were tested on the dedicated tester were also tested in a full speed, full load condition string test. One piece of data that can be extracted from the string test and correlated to the dedicated test rig results is the metal temperature of the bearings. The string test condition of 105 percent speed and 100 percent load yielded temperatures of 206.6°F (97°C) and 204.8°F (96°C) for the offset halves and TPJ bearings respectively. This correlates well to the test data collected on the dedicated tester with the temperatures being 208.4°F (98°C) and 213.8° (101°C) for the offset halves and TPJ bearings. These minor temperature discrepancies, less than 9°F (5°C), can be easily accounted for with variations in manufacturing and testing conditions.

CONCLUSIONS

Efficiency and oil flow are becoming a top priority for high performance gearboxes. Bearing selection can make a substantial difference in a gearboxes performance. This study shows that each bearing design tested has its strengths and weaknesses. Careful design and selection must be made by the engineer based on the desires and requirements of the end user. At lower journal velocities the pressure dam bearing will require the least oil flow but give the highest bearing temperature while maintaining a good power loss. The most demanding operating conditions require a TPJ to handle the loads and speeds but needs the most oil flow. The offset half bearing seems to be a nice compromise of the three bearings being able to handle most of the high speeds and loads. It has slightly more power loss but requires less oil than the TPJ. Bearing selection is no easy task, but with the help of this study the best bearing design for the operating conditions and customer needs can be selected.

NOMENCLATURE

T_{in}	- Oil inlet temperature
T_{drain}	- Test bearing oil drain temperature
T_{max}	- Embedded RTD temperature
Ns	- Rotational speed
Lu	- Projected area load
Q	- Test bearing oil drain flow
F	- Force applied to bearing
D	- Journal diameter of bearing
W	- Width of bearing
γ	- Weight per unit volume
Cp	- Specific heat of oil
Q	- Volumetric flow rate
ω	- Shaft rotational speed

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ACKNOWLEDGEMENTS

The authors would like to thank the Lufkin Industries test stand and its operators Michael Reep, Jamie Palmer and Eli Luna for their dedication in setting up and performing all of the bearing tests presented in this study.